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DESIGN AND ANALYSIS OF ROLLER SHAFTS FOR SUGAR CANE MILLS BY USING FEA TECHNIQUE WITH DIFFERENT PARAMETERS

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ABSTRACT

The scope of this paper is to analyse the strength and hardness of the sugarcane roller shaft, which has the main contribution in the sugarcane industries. Forged steel is used for making sugarcane roller shaft, in that, the fatigue level at the pinion end of the shaft causes more shearing effect with high fatigue level, so to eradicate these faults, the roller shafts was designed and analyzed. This paper deals with minimizing or reducing the effect of fatigue on the roller shaft. For minimizing the fatigue level, the following parameters are considered, they are; over loading of bagasse, due to over loading of sugarcane in the cane milling, and the pressure is increased. The increased pressure is due to materials like iron or steel that is too big and too hard, so that, there is high friction between the gun metals with the shaft. Lubrication and cooling is insufficient due to increase in pressure between the rollers. The performance of the roller shaft was fully analyzed for eliminating the shearing of roller shaft, combination of materials, change in deformation, minimizing the erosion due to friction and thermal distribution. For analyzing the strength of the material, load bearing capacity, material deformation, stress and thermal distribution in the roller shaft, ANSYS software is used and for designing PRO – E software is used. When the fatigue affecting factors are reduced, the aesthetic value of roller shaft is in good level.

KEYWORDS: Fatigue, Pinion, Shear Stress, Thermal Stress & Shaft and Roller

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Original Article

INTRODUCTION

In our day to-day life, sugar plays a vital and major role. It's the main ingredients for the majority of our food products. In India, refined sugar is essentially produced from the harvesting and crushing of the sugarcane. The harvested sugarcane is fed into the series of squashing rollers, and the sugarcane juice is extracted from the sugarcane for processing and refinement.

METHODOLOGY [THEORETICAL ANALYSIS]

In this study, a roller shaft is decided for design and analysis, and they are known for shaft diameters. Shell has a shrink fitted on the shaft. The shaft was supported by two journal bearings and the drive is directly mounted on the shaft. The roller is subjected to bagasse pressure and its rotational bending, torsion, twisting, direct shear and shrink fit stress. A pinion drives a base roller; sugarcane gets squashing between the top and bottom rollers. For designing the shafts of two roller mills, few assumptions are as follows:

- Force analysis on the roller shaft,
- Taper has less stress concentration factor than fillet.

SHAFT DIMENSION

Roller crush capacity = 2945.194tons/day, Power (P) = 474.85KW,

Roller speed (N) = 4 rpm Roller diameter = 920mm,

Shaft diameter at roller= 500 mm, Torque =1133KNm,

Shaft diameter at bearing support= 430 mm, Shaft diameter at pinion = 470 mm,

Self-weight = 13.1Ton, Total length = 4430mm

L1= 475 mm, L2= 1700mm, L3= 520 mm, L4= 271 mm, L= 2650 mm

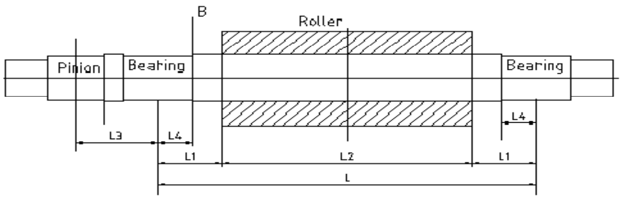


Figure 1: Dimension of Roller Shaft

FORCE ANALYSIS ON SHAFT

ASSUMPTIONS

- Friction loss in the bearing and gears are negligible.
- The gears mesh at pitch circles.
- The gear teeth have standard involutes tooth profiles.
- The effect of the dynamic forces is neglected.

PINION

The top roller driven by the motor, and also transmit the power to the other three rollers through pinion. The pinions are having same diameter. Due to pinion rotation, the loads acting on the shaft are shown in fig 2.

Outer diameter =984mm Number of teeth =16

Module =53mm Pressure angle =14.5°

Pitch circle diameter =848mm.

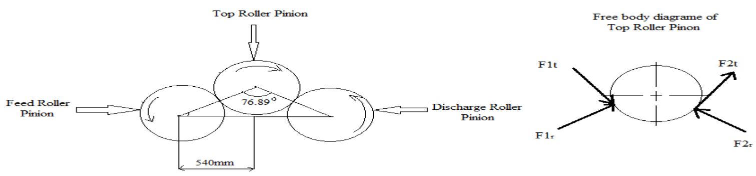


Figure 2: Pinion Forces

Tangential Force (F_t):

$F_{1t} = F_{2t} = P/V = 2673.634KN$.

Radial Force (F_r):

$F_{1r} = F_{2r} = F_{1t} x \tan ø = 691.448KN$.

Resolving Forces

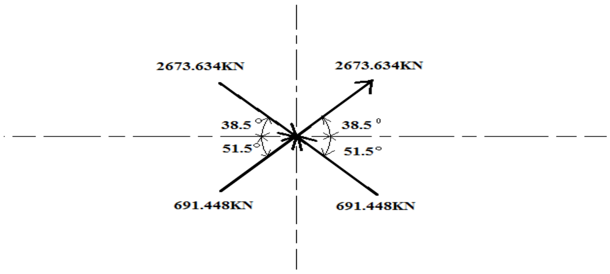


Figure 3: Free Body Diagram of Pinion

Horizontal Force: (→ +) ΣH = 4184.82KN

Vertical Force: (↑ +) ΣV = 1082.266KN

The self-weight of pinion is 25000N. It acts downwards, so it has '–' sign, so subtract from vertical load we get,

ΣV = 1057.266KN

SPROCKET

The top roller right end, sprocket is placed to transmit power from top roller to under feed roller. Due to the power transmission, the load is created on the spindle of the shaft as shown in fig 4.

Outer dia = 1171.2mm, No. of teeth = 17, Module = 60mm,

Pressure angle =14°, Pitch circle dia = 1020mm.

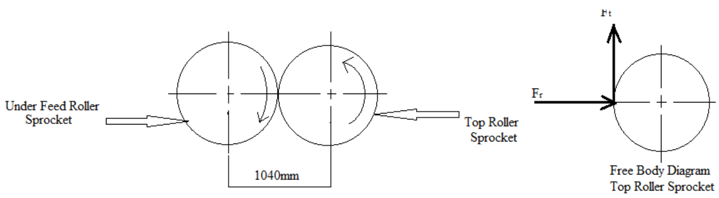


Figure 4: Sprocket Forces

Tangential Force (F_t); $F_t = P/V$, ΣV = 2222.786KN,

Radial Force (F_r); $F_r = F_t x \tan ø$, ΣH = 554.203KN.

SHELL

Shell is a grooved cylinder, which is mounted on shaft center. It crushes the bagasse fiber to extract the juice. This process required high force, which result more reaction force acting on the shell as shown in fig 5. Roller shell dimension of different roller, as shown in table 1

Table 1: Shell Dimensions

Rollers	Feed	Top	Discharge
Outer Diameter (mm)	878	915	900
Groove Depth (mm)	77.8	48	61.5
Pitch Circle Diameter (mm)	800.2	867	838.5

Setting: Feed roller side =36mm, Discharge roller side =14.5mm, Trash plate side =70mm

Center distance: Feed roller to top roller =869.6mm, Discharge roller to top roller =867.25mm

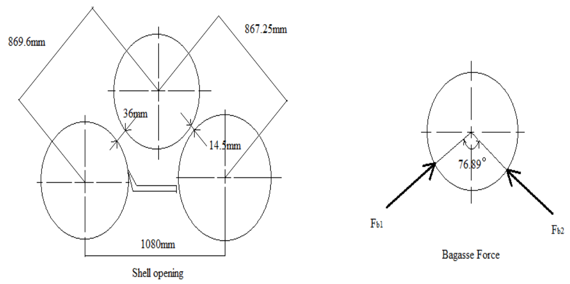


Figure 5: Shell Forces

BAGASSE FORCE AT FEED ROLLER SIDE

Due to the high compression of bagasse fiber, the roller can lift upward against the oil pressure. This can be neglected because of get high load on roller shaft.

Volume generated by shell openings = r x e_Ax υ x L (1)

Where, r- Re absorption factor,

e_A- Opening = 36mm, υ- Peripheral speed, L- Length of roller = 850mm.

Roller velocity (V) = πDN = 11.56m/min.

Re absorption factor(r) = 1.06+0.017V= 1.256

Peripheral Speed (υ) = V/60 = 11.56/60 = 0.193m/s

From (1) = $7.417\text{x}10^{-3}\text{m}^3$/s [Without Lifting]

Density (d_B) = weight/volume=0.034/$7.417\text{x}10^{-3}$

[Crush capacity=0.34tons/sec] =4.584 Tons/m^3

FORCE ON THE TOP ROLLER DUE TO BAGASSE COMPRESSION

F_{b1} =1300xLxDx$\sqrt{\epsilon_A}$xd_Bx12x (1+√r-1) (2)

Where, L-Length of roller = 850mm, D-Diameter of roller = 920mm,

ϵ_A-specific opening =e_A/D = 0.039

From (2) F_{b1} = 166.1MN/m

BAGASSE FORCE AT DISCHARGE ROLLER SIDE

Volume generated by shell openings = $r x e_A x \upsilon x L$ (3)

Where, r- Re absorption factor, e_A- Opening = 14.5mm, υ- Peripheral speed, L- Length of roller = 850mm

Re absorption factor(r) = 1.06+0.017V = 1.256

Peripheral Speed (υ) = 0.193m/s

From (3) =$2.987x10^{-3} m^3/s$ [Without Lifting]

Due to feed roller crushing weight loss on the bagasse fiber is 2/3.

So weight =0.0226 Tons/sec

Density (d_B) = 7.566 Tons/m^3

FORCE ON THE TOP ROLLER DUE TO BAGASSE COMPRESSION

F_{b2} =1300xLxDx$\sqrt{\epsilon_A}$xd_Bx12x (1+$\sqrt{r}$-1) (4)

Where, L-Length of roller = 850mm, D-Diameter of roller = 920mm, ϵ_A-Specific opening=e_A/D = 0.0157.

From (4) = F_{b2} = 174.155MN/m.

RESOLVING BAGASSE FORCE

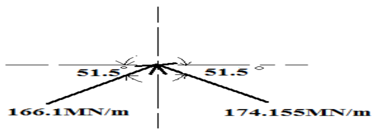


Figure 6: Free Body Diagram of Shell

Horizontal Force: (→ +) ΣH = -50.143MN/m

Vertical Force: (↑ +) ΣV = 266.286MN/m

RESULTANT LOADS

The above calculated loads are in different position at various locations; it can be resolved by two components.

i) Vertical Loads ii) Horizontal Loads

VERTICAL LOADS

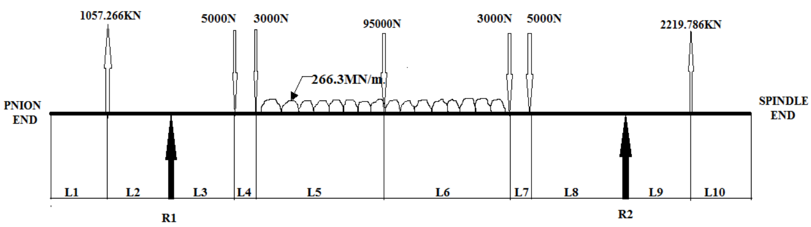


Figure 7: Vertical Loads on Shaft

L1=157.5mm; L2=487.5mm; L3=404.36mm; L4=37.4mm; L5=881.26mm; L6=881.26mm; L7=37.4mm; L8=405.36mm; L9=620mm; L10=515mm.

REACTION FORCES

Horizontal force (→ +) ΣH = 0, $R_1+R_2 = 0$

Vertical force (↑+) ΣV = 0, $R_1+R_2 = -455.876MN$ (5)

Moment about support 1(clock wise +), ΣM=0

R_2= -288.759MN, R_1= -227.12MN

HORIZONTAL LOADS

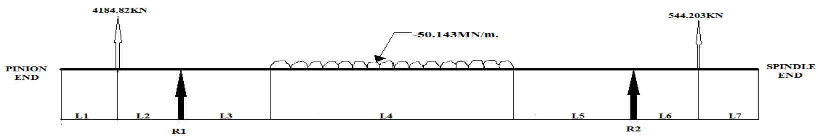


Figure 8: Horizontal Loads on Shaft

L1=157.5mm; L2=487.5mm; L3=475mm; L4=850mm; L5=476mm; L6=488.5mm; L7=515mm.

REACTION FORCES

Horizontal force (→ +), ΣH = 0

$R_1+R_2 = 80.514MN$ (6)

Vertical force (↑+), ΣV = 0

$R_1+R_2 = 0$

Moment about support 1(clock wise +), ΣM=0

R_2= 31.36MN, R_1= 49.154MN

FEA ANALYSIS OF ROLLER SHAFT

FINITE ELEMENT METHOD (FEA)

Finite element method is a numerical method for solving problems in Engineering and mathematical physics. In this method, a structure or a body in which the analysis can be carried out, and it's subdivided into smaller elements called finite elements. The assemblages of these elements are connected at a finite number of joints called 'Nodes' or Nodal points.

Steps for the FEA

- **Preprocessing**.

  a) Create the analysis design; b) Mesh the design.

- **Analysis.**

  a) Define material behavior, b) Apply loads & degree of freedom.

- **Post Processing.**

  a) Solve the design, b) Get results.

STRESS CALCULATION AT FILLET AREA

ANSYS is a tool used for finite element method; The ANSYS finite element program was used for finding stress distribution of top mill roller shaft. Key points were first created and then, line segments were formed. The lines were combined to create an area. Finally, this area was extruded and a three-dimensional circular roller shaft model was obtained, as shown in fig 9.

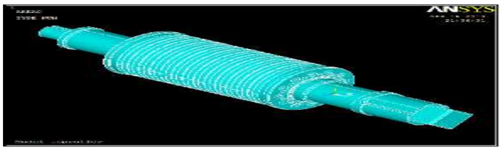


Figure 9: ANSYS Design of Roller Shaft

ELEMENT PROPERTIES

Analysis type: Structural

Element type: SOLID 45

Material properties for C-40 forged steel: i) Young's modulus =$2.05x10^6$ N/mm^2,

ii) Poisson's ratio= 0.3

MESHING

Meshing is the process of subdivided into smaller elements of shaft to obtain accurate result. Fine meshing also give high accurate result. The mesh element 3D tetrahedral elements were chosen and mesh, as shown in fig10.

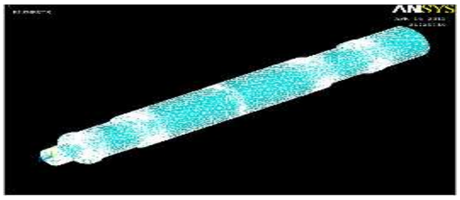


Figure 10: Fine Mesh View of Roller Shaft

LOADS

The above calculated loads are applied on the meshed roller shaft and solve. The result matrices are calculated for each element and formed the global matrix, to get the result of whole shaft. Due to the various loads acting on the shaft, the maximum stress occurred places were identified with the help of result.

FEA RESULT SHOULDER FILLET

In the survey, it is found that most of the failure occurs at the inner fillet side of the square end shaft. So, the stress can be identified on inner fillet side. The result could be simulate by the 'Von Mises Stress' theory.

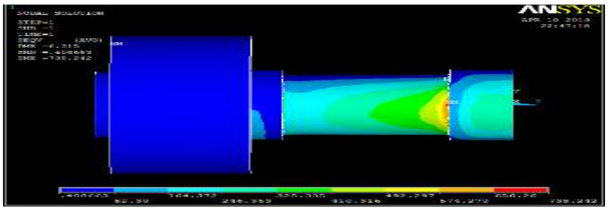


Figure 11: Stress Distribution at the Shoulder Fillet

The pinion end can only be focused for simulation. From the above result, due to the 15mm shoulder fillet radius on the journal portion, the maximum stresses were created, as shown in fig 11.

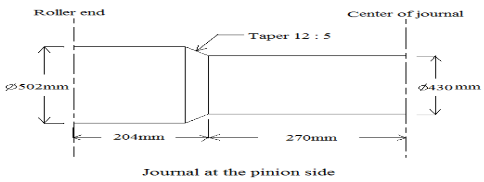
Maximum stress at inner fillet side=738.242MN/mm^2

TAPER SECTION ANALYSIS

Instead of providing fillet radius, the taper section can be used to reduce the stress at the journal portion. The angle of the taper increases, the stress will be reduced.

Taper ratio = (D-d)/L =12/5

Where, D – Major diameter = 502mm, d – Minor diameter = 430mm, L – Taper length = 30m. The selected taper ratio is 12:5.By using the above taper ratio, the shaft can be modified as shown in fig 12.



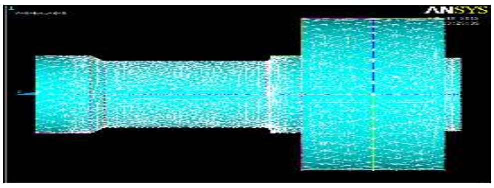


Figure 12: Dimension of Taper **Figure 13: Fine Mesh View of Taper Shaft**

FEA RESULT FOR TAPER SECTION

For the above taper ratio created instead for fillet radius, the same material properties and same load can be applied on the shaft. The result for the taper section is as shown in fig 14.

The maximum stress from the above result is due to the taper with ratio of 12:5 on the journal portion, as showed in fig 14. Maximum stress at taper side=525.395MN/mm^2

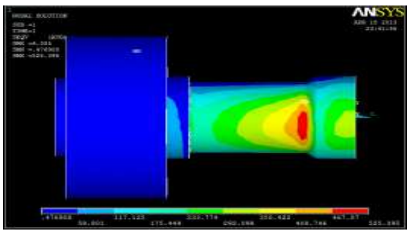


Figure 14: Stress Distribution in the Taper Shaft

RESULTS AND DISCUSSIONS

The top mill roll shaft is analyzed theoretically, as well as with the help of FEA software for its safe working, by checking various parameters within limits. Consider the above two methodologies to determine the stress distribution

factor of the first case, with shoulder fillet radius of 15mm was found out as 738.242 MN/mm^2.

In the second case, tapered fillet with the taper ratio of 12:5 the stress distribution was found as 525.395 MN/mm^2. The stress distribution on the shaft in the second case is decreased by an amount 212.847 MN/mm^2.

Stress distribution reduction factor using FEA results = 212.847/738.242

= **28.83%**

CONCLUSIONS

The most of the failures occur on inner fillet side at square end of the top mill roll shaft. The stress on fillet can be analyzed, which has one of the major factor for failure. The fillet can be changed as taper, so that the stress on the shaft can be reduced. This can be proved theoretically by using ANSYS software.

As per simulation result, we found that the stress distribution for shoulder fillet is more, when compared to stress distribution for taper fillet on the pinion end of the roller shaft, as tabulated in table 2.

Table 2: Result

Cases	FEA Stress (MN/mm^2)
Shoulder fillet	738.242
Taper section	525.395

From the above result, we concluded and suggested that by using tapered fillet instead of existing shoulder fillet, radius would reduce the stress and simultaneously increases the reliability of the shaft by **28.83%**.

REFERENCES

1. *H. Gao and L. G. Qu.(2002), Department of Mechanical Engineering, Shenyang Institute of Aeronautical Engineering, Shenyang 110034, PR China, '3D design and analysis of the crushing roller of a high-pressure grinding roller', Vol 129,issue 1-3, pp 649-652.*
2. *J.S. Rivas, J.J. Cornado, A.L. Gomez,(2006), Mechanical Engineering School, Universidad dell Valle, Cali, Columbia, 'Tribological aspects for the shafts and bearings of sugar cane mills', J.S. Rivas et al. / Wear 261 (2006) 779–784.*
3. *S.L.A. Rodriguez, C.A. Volez.(2005), Mechanical Engineering School, Universidad del Valle, Colombia 2Manuelita S.A.Mill, Colombia, 'Increasing the reliability of the top roller shafts in sugar mills', Proc. ISSCT, Vol. 25. 2005.*
4. *N. Arzola, S. A. Rodriguez, J.J. Cornado.(2003),118p,Mechanical Engineering School, Universidad del Valle, Cali Colombia, 'Life prediction for the top roller shafts of the sugar mills'.*
5. *Karthi.R.R[1], Dhanabalan.S[2] Comparative analysis of Plain and Herringbone Grooved Journal Bearing Under The Hydrodynamic Lubrication Conditions In Pak. J. Bio technol. Vol. 14 Special Issue I (International Conference on Futuristic Innovations in Mechanical Engineering and Manufacturing Management) Pp. 25- 31 (2017).*
6. *Karthi, R.R[1]., Dhanabalan.S[2], An Investigation on Micro Robot for Medical Applications in Pak. J. Bio technol. Vol. 14 special issue I (International Conference on Futuristic Innovations in Mechanical Engineering and Manufacturing Management) Pp. 32- 36 (2017).*
7. *Dhanabalan.S[1], Karthi.R.R[2], Sivakumar K[3], and Sathiya Narayanan C[4] Optimization of rotary EDM Process Parameters for Inconel 718 Using Artificial Neural Network in Pak. J. Bio technol. Vol. 14 special issue I (International Conference on Futuristic Innovations in Mechanical Engineering and Manufacturing Management) Pp. 58- 60 (2017).*

8. *Jay G Patel et al.,, To Study and Optimize the Process Parameter Throw Roller Burnishing Process on EN19 Material by Response Surface Methodology, International Journal of Mechanical and Production Engineering Research and Development (IJMPERD), Volume 6, Issue 6, November - December 2016, pp. 21-30*
9. *Dhanabalan.S[1], Karthi.R.R[2] Multi-Objective Optimization of EDM Parameters for Ti alloy in Pak.J. Bio technol. Vol. 14 special issue I (International Conference on Futuristic Innovations in Mechanical Engineering and Manufacturing Management) Pp. 54- 57 (2017).*